with the distance piece. Other makers prefer flat guides cast with the frame, which gives a solid backing to take the thrust due to the obliquity of the connecting-rod. A point in their favour is that the working parts are more open for inspection and adjustment, and when the doors are off, such parts are as accessible as in an open engine. In the case of the bored variety, openings are, of course, made at the sides to give accessibility.

Connecting-rod.—In high-speed engines the ratio of length of connecting-rod to radius of crank is from 5 to 7. The longer the rod the less is the disturbing influence of the inertia forces at the end of the stroke, and the pressure on the guide is reduced owing to the obliquity of the rod being less. At the same time the height of the engine is increased, and this is sometimes an important consideration where head room is restricted. The design of the top end of the rod is determined by the type of crosshead used. In that type in which the crosshead pin is fixed usually by shrinkage in the jaws of the rod, the width of the jaws may be 0-6^ + 0-2,

and the thickness of metal round the eye - + -J- in., where d is the dia-
3

meter of the pin. The depth of the jaw must be sufficient to clear the nuts on the crosshead bolts. The sides of the jaws are usually made a straight taper. The diameter of the rod at the junction with the jaws may be a little less than the diameter of the piston-rod, and the body of the rod increases in diameter by a straight taper to the foot.

In the other design in which the gudgeon-pins form part of the crosshead, the bearings are carried by flat palms formed on the fork of the rod. There are thus two bearings, the length of each being about the same as or 25 per cent more than the diameter, and the total area of these bearings must be such as to give a pressure of from 1000 to 1200 lb. per square inch. This figure also applies to the design first described, in which the crosshead pin is fixed in the jaws of the connecting-rod. The total area of the bolts at the bottom of the thread may be such as to give a stress of 3500 to 4000 lb. per square inch. The width of the caps and of the

palms forming the rod may be about 0-7 of the length of the bearings, and their thickness may be the same as or a little greater than the diameter of the bolts. The stress upon them due to bending should be checked, and should not exceed

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8000 Ib. per square inch. The bending moment should be taken as —,

where L is distance between bolts and P the total pressure on high-pressure piston. The steps as in the case above described, that is, when incorporated with the crosshead, should be of phosphor bronze or hard gun-metal. The thickness of metal in the sides of the fork should be such that the stress due to the piston load should not exceed 9000 Ib. per square inch, when calculated upon a basis of combined direct tension and bending. This should be checked at several sections.

The bottom end is invariably of the marine type. In smaller engines the steps are made of gun-metal lined with white metal, but in medium-size and larger engines the steps are made of cast steel also lined with white